SUMMARY REPORT

#360

Battelle Memorial Institute

505 KING AVENUE COLUMBUS 1, OHIO

June 10, 1957

Officer in Charge
U. S. Naval Civil Engineering
Research and Evaluation Laboratory
Port Hueneme, California

Attention Commander C. J. Merdinger, CEC, USN

Dear Sir:

Sea-Water Distillation Project Noy-73219

Enclosed are six copies of a summary report on "The Development of Evaporators for Advance-Base Thermocompression Sea-Water Stills". An additional 119 copies are being forwarded to you under separate cover.

This summary report covers the period November 15, 1955, to March 15, 1957, during which the research effort was directed exclusively toward the improved design of evaporators of sea-water stills. A previous summary report dated November 15, 1955, presented the results of evaluations of the other components of thermocompression stills.

Our studies have shown that the rate of heat transfer in evaporators of the configuration presently used in advance-base stills can be improved appreciably by means of forced circulation of the evaporating water and by means of dropwise condensation of the steam. However, it appears that only the gain resulting from dropwise condensation can be utilized to improve the economy of operation. The gains in heat transfer resulting from forced circulation, although extremely effective in decreasing steam compression power are offset by the additional power required for circulating the evaporating water.

Although the studies have shown that substantial improvement in heat transfer can be effected in advance-base stills, the work is by no means complete. Time did not permit investigating several areas of potential improvement which appear attractive, and which should be pursued further. These are outlined at the end of the report.

Yours very truly,

James A. Eibling

James A. Eibling

JAE:jpl

SUMMARY REPORT

on

THE DEVELOPMENT OF EVAPORATORS FOR ADVANCE-BASE THERMOCOMPRESSION SEA-WATER STILLS

INTRODUCTION

This report presents a summary of the work performed from November 15, 1955, to March 15, 1957, on the development of advance-base thermocompression sea-water stills. During this period the research effort was directed solely toward the improvement of evaporators for thermocompression stills. The work was carried out on an extension of a previous research program which began in April 1953 and which covered evaluations of all of the components of thermocompression stills. The results of the previous research program are contained in a summary report dated November 1955*.

SCOPE AND OBJECTIVES

The earlier studies showed that, of the several components that comprise a thermocompression still, the evaporator is probably the most important from the standpoint of obtaining high over-all performance of a still. Accordingly, in an effort to determine what improvements could be made in evaporator design, fundamental studies of heat transfer from condensing steam to boiling water were undertaken. All of the studies were concerned with improvements that could be made in conventional vertical shell-and-tube-type evaporators. Unusual or radically different types of equipment such as a rotary-film-type evaporator were not studied in this research program.

In determining to what extent improvements could be made some standard for comparison was necessary. For this purpose, the performance and characteristics of a hypothetical evaporator believed to be representative of the best current practice was used. This hypothetical evaporator, which is described in detail in the previous summary report would have a heat-transfer surface made up of 650 vertical, 5/8-in. OD 18 BWG tubes, 36 in. long. Operating on sea water at a pressure difference of 4-in. Hg, the evaporator would produce 90 gph of distilled water and would have an over-all heat-transfer coefficient of 525 Btu/(hr)(ft^2)(F) with natural convection evaporation and film condensation.

In approaching the objective of the development of an improved evaporator, it is important to recognize that the attainment of high rates of heat transfer is not the sole criterion on which to base performance evaluation. Inasmuch as a thermocompression still operates on a partially closed cycle, the effectiveness of each major component is,

* Summary report to U. S. Naval Civil Engineering Research and Evaluation Laboratory, "The Development of Advance-Base Thermocompression Sea-Water Stills" from Battelle Memorial Institute, November 1955.

ranging from 0.65 to 10 fps and with both film and dropwise condensation. Vertical brass tubes approximately 3 ft long of 1/2-in. and 3/4-in. OD and 16 BWG were used in the tests.

In the tests of dropwise condensation, the tubes were coated with a Teflon film approximately one-half mil thick. Teflon has a particularly desirable feature in this application in that it presents a permanent dropwise promoting surface, whereas other materials used in previous investigations wash away and must be continually replaced. The Teflon film offers some resistance to heat flow but the net effect is to increase heat transfer significantly. For example, at condensing film temperature differences usually encountered in thermocompression stills, tests showed that the film coefficient was about twice as much for dropwise condensation as for film-type condensation. The increase in condensing-film coefficient effected an improvement of up to 50 per cent in the over-all heat-transfer rate, depending on the flow rate of the evaporating water and the over-all temperature difference used. The greatest gains occurred with a combination of small values of Δt and high flow rates. For example, with a Δt of 2 F and a flow velocity of 10 fps, dropwise condensation gave a 50 per cent increase over film condensation. With natural convection evaporation and a 4 F Δt, dropwise condensation gave an increase in heat transfer over film condensation of about 30 per cent.

The results of the forced-convection tests show that the over-all heat-transfer rate can be doubled by increasing the velocity of the evaporating water from 3 fps to 10 fps. A velocity of 3 fps is believed to be near the upper limit obtainable with natural convection with 2 fps or less probably being more typical.

On the basis of the results of the heat-transfer studies, two improved evaporator designs for a 90-gph still are presented. The first design would effect an improvement in the performance factor of a still of about 23 per cent over the best known previous design with only a ten per cent increase in heat-transfer surface. The second design would produce a performance factor of 300 lb distillate per lb fuel, which is the same as the best known previous design but would require about 20 per cent less heat-transfer surface. In both designs, only the increase in heat transfer due to dropwise condensation would be utilized.

In spite of the fact that forced-convection evaporation gives large increases in heat-transfer rates, forced convection operation does not appear to be advantageous for use in advance-base thermocompression stills designed to operate with low Δt's, that is less than 10 F. The experimental study shows that with low Δt, the pumping power required for forced convection circulation is not compensated for by the reduction in steam compressor power associated with the improved heat transfer resulting from the forced convection. Thus, the total power input to a thermocompression still is greater with forced convection than with natural convection, which, of course, results in a lower performance factor. The trend of the data indicates that at over-all temperature differences higher than those deemed practical for an efficient thermocompression still, forced-convection evaporation would be beneficial. With certain other types of equipment, for example multiple-effect evaporators, forced convection would probably be of considerable benefit.

A section of this report gives a summary of material obtained in a survey of substitute materials that might be used in the fabrication of sea-water evaporators. The information obtained in the survey shows that few data are available on sea-water corrosion at the temperatures and velocities encountered in thermocompression stills.

Based on present knowledge and experience, a 70-30 copper-nickel alloy is recommended for evaporator tubes, as well as for the tubes in the other heat exchangers of a still. The evaporator designs presented in this report use this 70-30 copper-nickel alloy for all the parts in contact with salt water. The suggestion is made of the long-range possibility of using solid titanium or titanium coated tubes for evaporators. Titanium is essentially immune to attack by sea water, and there is evidence that titanium presents a surface which, to some degree, induces dropwise condensation without any additional promoting agent. With regard to aluminum, considerable research would be needed to demonstrate the possibility that useful life could be obtained with an all-aluminum design.

GENERAL DISCUSSION OF HEAT TRANSFER IN EVAPORATORS

A brief review of the heat-transfer processes occurring in the evaporator of a thermocompression still is presented here in order to point out the areas in which improvement can be expected and to show how the results of the research program support the objective of developing an improved evaporator.

Heat transfer in an evaporator of a thermocompression sea-water still takes place between condensing steam surrounding the evaporator tubes and boiling sea water inside the tubes. The total resistance to heat flow across a tube is made up of the sum of four separate resistances: (1) the resistance of the layer of condensed steam, (2) the resistance of the tube wall, (3) the resistance of scale inside the tube, and (4) the resistance of the evaporating film inside the tube. The characteristics of each of these resistances are described in the following subsections.

Condensing Film

Two modes of condensation are known: film and dropwise. Film condensation occurs on a wettable cooling surface. With film condensation, a continuous layer of condensate covers the tube surface and flows down the tube under the influence of gravity; it is immediately renewed by further condensation. The thickness of the film increases as the flow moves downward in proportion to the height of the tube. The latent heat released by condensation passes through the film from the condensing vapor to the tube wall. The film offers considerable resistance to heat flow because of the low thermal conductivity of the liquid. Therefore, there is an appreciable temperature drop across the film. Nusselt's theoretical equation for film-type condensation shows that the rate of heat flow across the tube per unit length of tube is proportional to the three-fourths power of the temperature difference between the condensing steam and the tube wall, and the rate of heat flow per unit area of tube surface is proportional to the three-fourths power of the height of the tube.

The second known mode of condensation is called dropwise condensation because the condensed vapor forms on the tube surface as separate drops. The drops of condensate coalesce and run down the tube surface. These are immediately replaced with new drops from the condensing vapor. Four conditions favor dropwise condensation. These are: (1) low rate of condensation, (2) low viscosity of condensate, (3) high surface tension of the condensate such that the cohesive force of the condensate is greater than

the force of adhesion between the condensate and the tube surface, i.e., the tube surface must be such that it is not wetted by the condensate, and (4) smoothness of the cooling surface.

Of these four conditions, the third is probably the most important, since dropwise condensation will not persist if the tube is wetted by the condensate, regardless of the other conditions. The condition of low condensation rate already exists in thermocompression stills because of the relatively small heat flux associated with the low over-all temperature differences used.

Dropwise condensation has, under some conditions, produced film coefficients four to eight times greater than for film-type condensation, primarily because the drops on the tube surface offer much less thermal resistance than a continuous film. Pure dropwise condensation will not persist on any of the present commercially available tube materials but partial dropwise condensation or mixed condensation can occur on copper tubes and to a lesser extent on copper-base-alloy tubes. In studies of pure dropwise condensation, previous investigators have coated tube surfaces with a fatty acid, mineral oil, or a mercaptan. In a practical application, however, these coatings soon wash away and must be continually replaced. Moreover, they may impart an undesirable odor or taste to the distillate. Thus, a need exists for a permanent-type dropwise promoter which will not deleteriously affect the distillate and which can be applied to the tube in an extremely thin coating to minimize the resistance to heat transfer.

In connection with the consideration of the resistance offered by the condensing film, mention should be made of the importance of purging noncondensable gases from the steam chest. The presence of even small amounts of noncondensable gases greatly retards the rate of heat transfer during condensation. Inasmuch as it is virtually impossible to prevent at least some noncondensable gases from entering the steam chest, continuous and effective venting must be resorted to in order to keep the concentration of noncondensable gases to a minimum.

Tube-Wall Resistance

The resistance of the tube wall to heat flow depends on the tube-wall thickness and on the thermal conductivity of the tube material. In some types of heat-transfer equipment, the resistance of the tube wall is so small that it is often neglected in approximate heat-transfer calculations. However, in the evaporator of a thermocompression still, where the over-all temperature difference is low, the tube-wall resistance cannot be considered to be negligible. In a thermocompression still the resistance to heat flow of the evaporator tube wall ranges between 1×10^{-4} and 3×10^{-4} $(ft^2)(hr)F/Btu$ as contrasted with an over-all resistance in the range of 10×10^{-4} to 20×10^{-4} $(ft^2)(hr)F/Btu$. Thus, the resistance of the tube wall is 10 to 30 per cent of the total resistance to heat transfer. At present, evaporator tubes are selected primarily for their corrosion resistance to salt water. The copper-nickel alloys which are generally used are relatively poor thermal conductors. If alloys with the necessary corrosion resistance and thermal conductivity similar to copper or brass were available significant gains in thermocompression still performance could be effected.

Scale Resistance

The scale deposited on the evaporator tubes from sea water evaporating at approximately atmospheric pressure consists principally of magnesium hydroxide with a smaller amount of calcium sulfate. The rate of scale deposition increases with increase in the rate and temperature of evaporation and with increases in brine concentration. Because scale deposits on the inside of evaporator tubes may increase the over-all resistance to heat transfer by as much as 25 per cent, or more, it is imperative to remove the scale periodically and to keep its formation to a minimum.

Evaporating Film

The mechanism of heat transfer to an evaporating liquid is complex and few data are available for low-temperature differences such as exist in evaporators of thermocompression stills. In a sea-water evaporator, water is usually introduced into the bottom of the tube section several degrees below the boiling point. As the water flows upward in the heated tubes, the temperature rises until the water is brought to the boiling point. Further heating results in boiling which may take place over a considerable portion of the tube height. However, because the pressure falls owing to wall friction and the reduction in the hydrostatic head, the evaporating temperature decreases with increasing height of the tube.

The evaporating film coefficient for water is dependent on the nature of the surface on which boiling takes place and on the degree of turbulence in the film. A steam bubble will not form in water unless there is a discontinuity to start the bubble. A rough surface, say one that has been sandblasted, serves this purpose and permits a higher heat flux than a polished surface. The roughened surface has many more nuclei for the formation of bubbles. In both forced and natural-convection evaporation in an evaporator tube, the flow is turbulent. In forced convection, however, the turbulence is greater and therefore the rate of heat transfer is higher. In addition to the turbulence created by the convection of the water, the formation and release of vapor bubbles in the boundary layer film agitates the film, thereby further increasing the rate of heat transfer. Also, as the bubbles pass upward through the main body of water they exert a stirring effect which serves further to increase the turbulence of the water.

Three separate zones of heat transfer usually occur in the evaporator tube. In the lowest zone in the tube, near the tube entrance, the flow consists of water and the film coefficient is the same as for water flow without evaporation. In the midportion of the tube where vigorous boiling is taking place, the rapid formation and release of the vapor bubbles from the tube wall results in higher rates of heat flux than for water without boiling. In the upper section of the tube, if all of the water has vaporized, the local film coefficient becomes less than it is in the boiling region because heat is being transferred from the tube wall to a gas. Heat-transfer film coefficients are of course much lower for gas films than for liquid films.

Concluding Remarks

In consideration of the four resistances to heat transfer, there are several ways in which improvements can be made. One, of course, is to use the thinnest possible tube material having extremely high thermal conductivity. Although some improvement could be made in this manner the gain would be small compared to those that appear possible by other means.

The problem of reducing scaling remains as one of the most important problems yet to be solved in sea-water evaporation. The best method now known for reducing scaling in advance-base stills appears to be the periodic injection of citric or other acid. Although this method is not completely effective, it does reduce the average thickness of scale appreciably and thereby enhances heat transfer. Research needed in the area of scaling is primarily one of basic studies of the chemistry and physics of scale formation.

The best possibilities for improving heat transfer that lie within the scope of this research program are those associated with reductions in the thermal resistances of the evaporation and condensation films. Included in this category are the specific items of (1) increasing the velocity of the evaporating water inside the tubes, (2) inducing dropwise condensation, and (3) the use of drip dams and extended surfaces on the condensing side of the tubes. The diameter and length of the tubes are also factors to be considered in optimizing the design of an evaporator.

The experimental program carried out at Battelle has been planned to give data needed to evaluate the effectiveness of some of these suggested methods of improvement. The experimental work completed has been chiefly concerned with forced-convection boiling at various water velocities and various temperature differences and with dropwise condensation. The results of this work are presented in the following sections.

TEST APPARATUS AND PROCEDURE

Description of Test Equipment

Figures 1 and 2 are respectively a schematic diagram and a photograph of the experimental equipment that was assembled to study heat transfer during evaporation and condensation. The apparatus was designed to provide over-all temperature differences between condensing steam and evaporating water in the range of 2 to 18 F at forced convection evaporating water flow rates ranging from 0.5 to 10 fps. The evaporator was sized to provide for testing of up to five 36-in. long tubes. However, the tests were actually made with one 3/4-in. OD tube, five Teflon coated 1/2-in. OD tubes, three 1/2-in. OD Teflon coated tubes, and three 1/2-in. OD plain brass tubes.

The condensing steam was generated in an electrically heated steam generator and was piped to the condensing side of the evaporator. The condensing steam pressure was controlled with a recording pressure controller which maintained the pressure within ± 0.1 psi of the set point. The heat input to the apparatus was measured with two watt-hour meters which could be read accurately to one watt-hour.

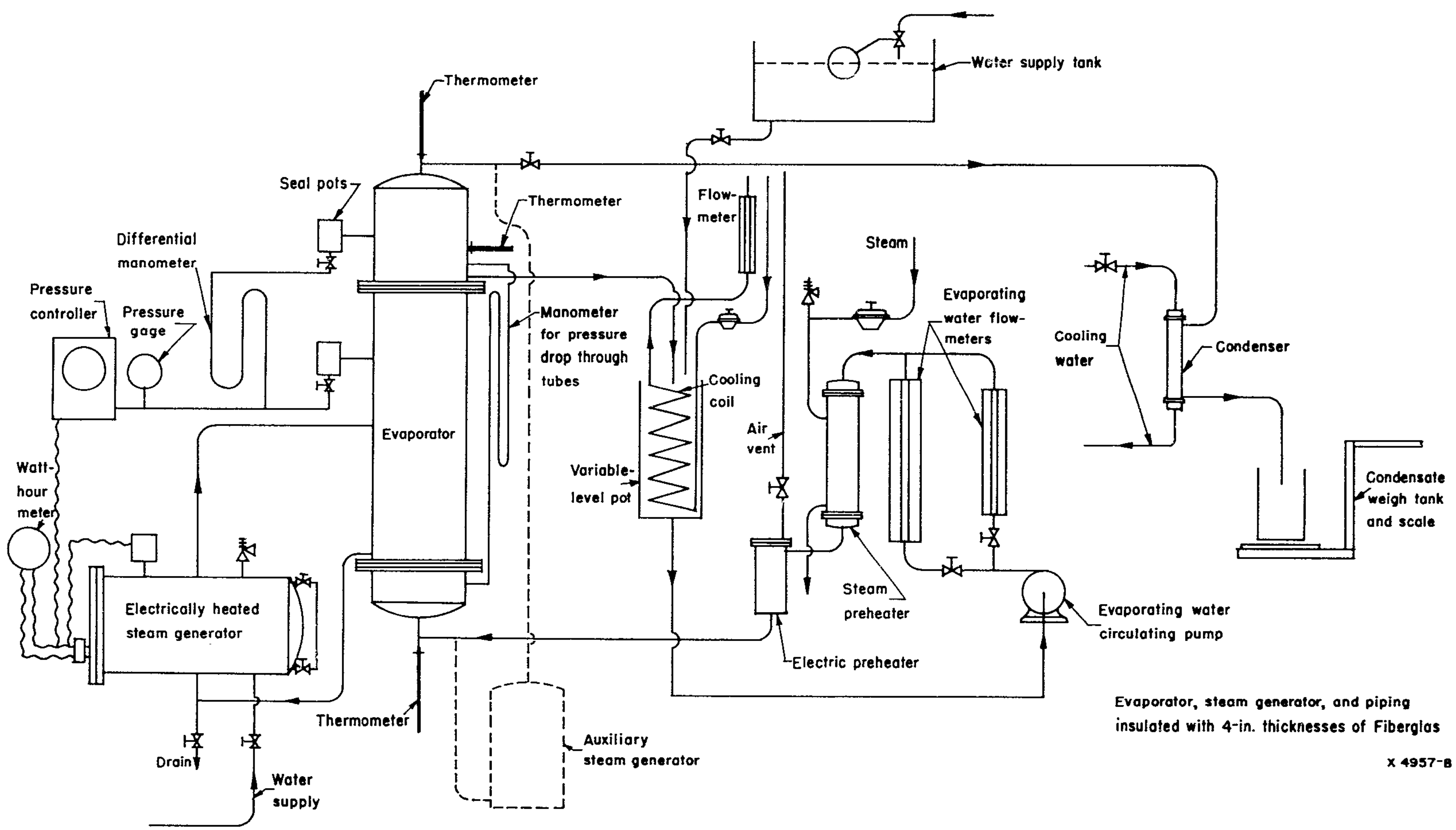
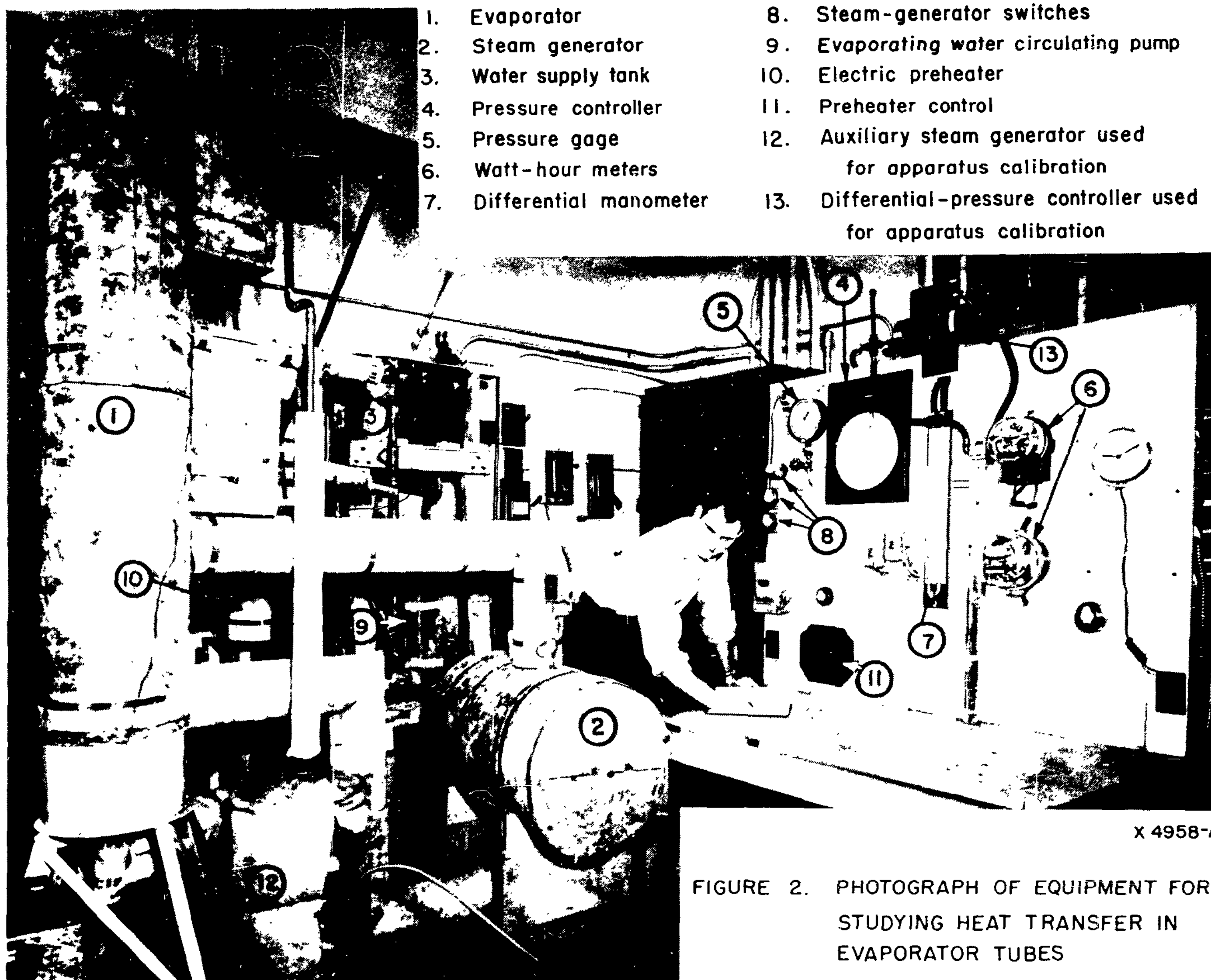


FIGURE 1. SCHEMATIC DIAGRAM OF EQUIPMENT FOR STUDYING HEAT TRANSFER IN EVAPORATOR TUBES



1. Evaporator
2. Steam generator
3. Water supply tank
4. Pressure controller
5. Pressure gage
6. Watt-hour meters
7. Differential manometer
8. Steam-generator switches
9. Evaporating water circulating pump
10. Electric preheater
11. Preheater control
12. Auxiliary steam generator used for apparatus calibration
13. Differential-pressure controller used for apparatus calibration

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FIGURE 2. PHOTOGRAPH OF EQUIPMENT FOR STUDYING HEAT TRANSFER IN EVAPORATOR TUBES

In order to circulate the evaporating water, an all-bronze centrifugal pump was installed in the circulating system in such a way that the pump discharge could be controlled by means of a throttle valve and by varying the speed of the pump. The evaporating water flow rate was measured with two rotameters in parallel. One was used for low flow rates, the other for high flow rates. A combination of electric and steam-heated preheaters was used in the circulating system to maintain the temperature of the evaporating water at the entrance to the evaporator tube within 1 F of the boiling point. The over-all temperature differences between the condensing and evaporating sides of the system was determined by converting pressure difference measured with a mercury manometer to temperature difference.

An effort was made to run natural convection tests. To run these tests, an open tank, in which the water level could be varied, was installed in the evaporating water circulating system in such a manner that the water level in the tubes could be controlled. Also the circulating pump was throttled so that the head developed equalled the friction losses in the external piping. The natural-convection experiments were not successful because the pump capacity was too small at the very low discharge heads required. It is believed, however, that this general method of simulating natural convection is workable. It has the advantage that the flow rates are easily measured, but the method requires a pump with low-head, high-capacity characteristics. A small axial-flow pump appears desirable for this application but none is commercially available.

Teflon-Coated Tubes

Teflon-coated tubes were used to promote dropwise condensation in one series of tests. Teflon was selected because it is one of the few dropwise promoting agents which is permanent. Most of the dropwise promoters used in the past, such as mineral or lard oil, must be periodically injected into the condensing steam. The Teflon film, used in the experiments was estimated to be on the average about 0.0005-in. thick; it was applied by hand spraying Du Pont's Teflon One-Coat Enamel No. 851-204 on the tubes and curing the enamel at 690 F for 1-1/2 min in a hot-air furnace.

A method of coating heat-exchanger tubes with an extremely thin, uniform coating of Teflon was developed at Battelle under the sponsorship of the Griscom-Russell Company of Massilon, Ohio. Although the coating techniques developed and the use of Teflon-coated tubes are considered to be of a proprietary nature, Griscom-Russell kindly gave permission for the use of Teflon-coated tubes in the present experimental work. It should be pointed out that the method employed for coating the experimental tubes produced a less uniform coating than would be possible with the refined technique now used by the Griscom-Russell Company. Thus, it is believed that higher experimental condensing coefficients would have been obtained if the refined technique had been used.

Calibration of Heat Loss of Test Apparatus

In order to determine the amount of heat actually transferred across the evaporator tubes, it was necessary to calibrate the apparatus for heat losses to the atmosphere. For this purpose, an auxiliary steam generator and differential pressure controller were connected to the apparatus to supply steam to the inside of the evaporator

tubes at the same pressure and temperature as the steam on the outside of the tubes. With no temperature difference across the tubes no heat was transferred. Therefore, all the heat put into the system, as measured by the watt-hour meters, was that lost to the atmosphere. A series of runs was made over a range of steam temperatures and a heat loss curve was plotted for Btu/hr loss versus temperature difference between condensing steam and room air.

Test Procedure

The general procedure followed in the tests was first to determine the over-all heat-transfer rate and then to determine the boiling and condensing film coefficients. The heat-flux was calculated from data obtained by measuring the heat transferred across the evaporator tubes, the temperature difference between the condensing steam and evaporating water, and the velocity and temperature of the evaporating water at the inlet to the tubes.

The condensing and evaporating-film coefficients were determined in the following manner. First the over-all coefficient was measured with hot but not boiling water inside the tubes and with steam condensing on the outside. The film coefficient for the hot water inside the tube was then calculated from the well-known Seider-and-Tate equation. The condensing coefficient was next determined by subtracting the calculated hot-water film coefficient from the over-all coefficient. Finally, the boiling-film coefficients were determined by subtracting the condensing film coefficient from the over-all coefficients measured with evaporating water inside the tubes.

RESULTS OF HEAT-TRANSFER TESTS

Three series of heat-transfer tests, A, B, and C, were made during the course of the project. All of the tests were conducted using the experimental, laboratory evaporator, with forced-convection boiling and with the over-all temperature difference between condensing steam and evaporating water in the range of 2 to 18 F. For the tests in Series A, one 3/4-in. OD, 16 BWG brass tube, nominal composition, 66.5 per cent Cu, 33 per cent Zn, 0.5 per cent Pb was used with forced-convection velocities ranging from 0.65 to 6 fps. Presumably, film-type condensation took place on the outside of this tube. The Series B tests were run using three 1/2-in. OD, 16 BWG brass tubes, nominal composition 68.5 - 71.5 per cent Cu, 0.075 max per cent Pb, 0.06 max per cent Fe, and the remainder Zn. For the Series B tests the tubes were coated with a film of Teflon approximately one-half mil thick to promote dropwise condensation. Forced-convection velocities of 3, 6, and 10 fps were used in these tests. Series C tests were identical to those of Series B except that Teflon was not applied to the tubes so that film condensation prevailed.

Over-All Heat-Transfer Data

Figures 3, 4, and 5 present, in curve form, the results of test Series A, B, and C respectively. In each figure, the relation of the heat flux to the over-all temperature

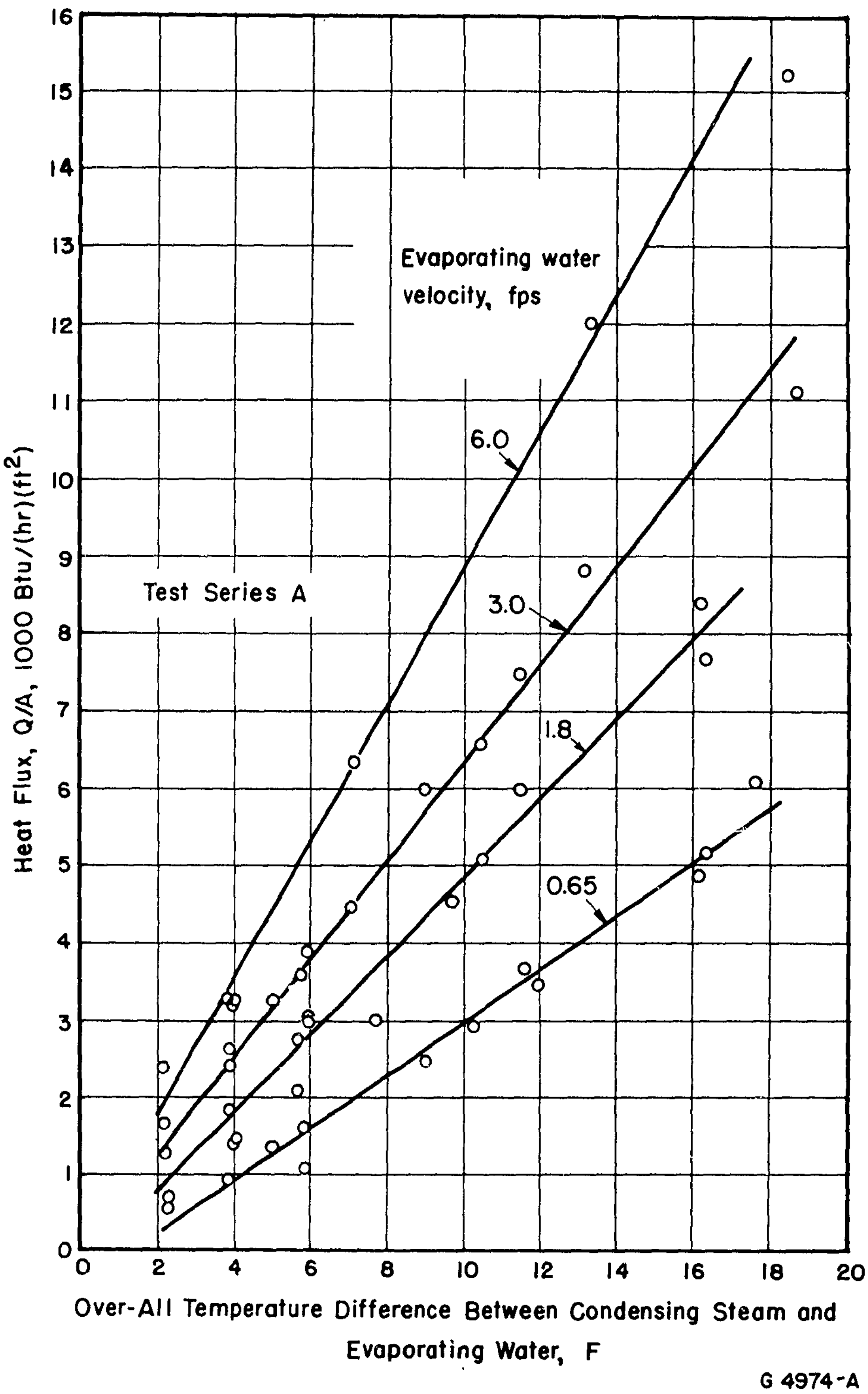


FIGURE 3. RELATION OF HEAT FLUX TO OVER-ALL TEMPERATURE DIFFERENCE AND EVAPORATING WATER VELOCITY FOR ONE 3/4-INCH-OD, 16 BWG BRASS TUBE, FILM CONDENSATION

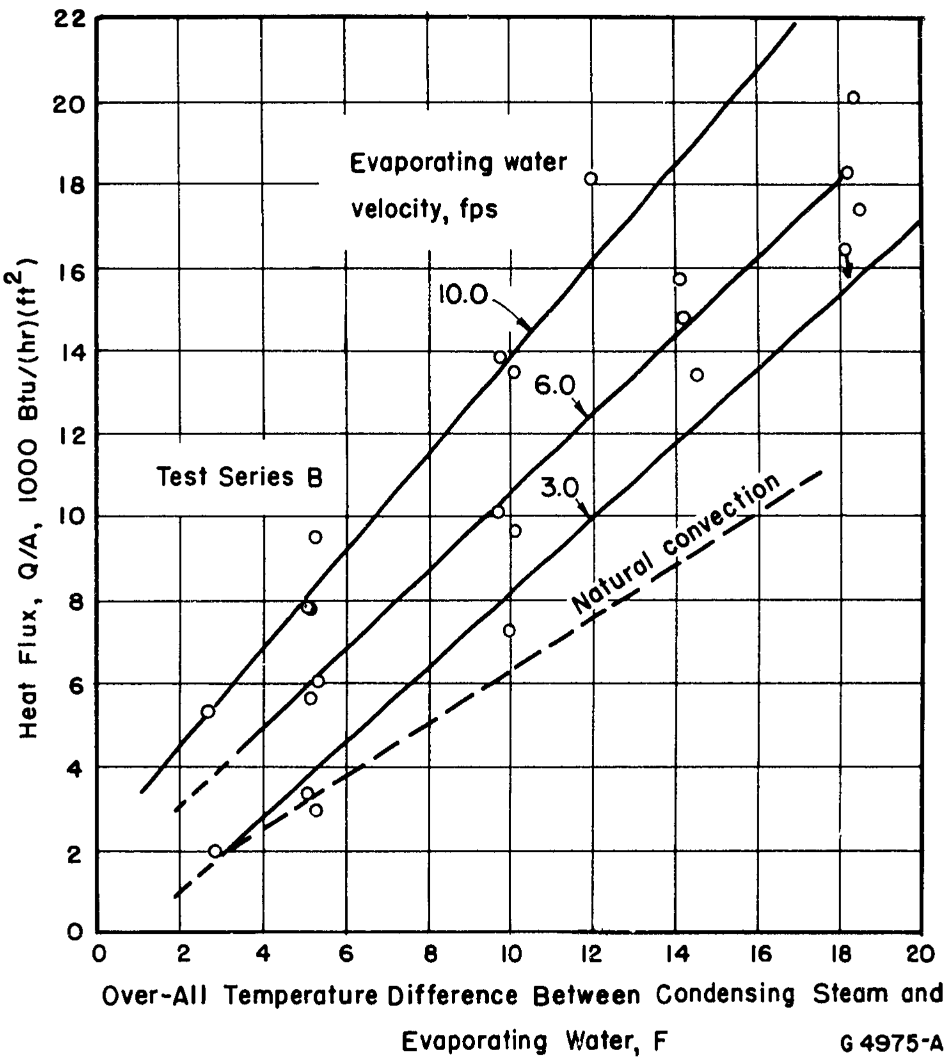


FIGURE 4. RELATION OF HEAT FLUX TO OVER-ALL TEMPERATURE DIFFERENCE AND EVAPORATING WATER VELOCITY FOR THREE 1/2-INCH-OD, 16 BWG TEFLON-COATED BRASS TUBES, DROPWISE CONDENSATION

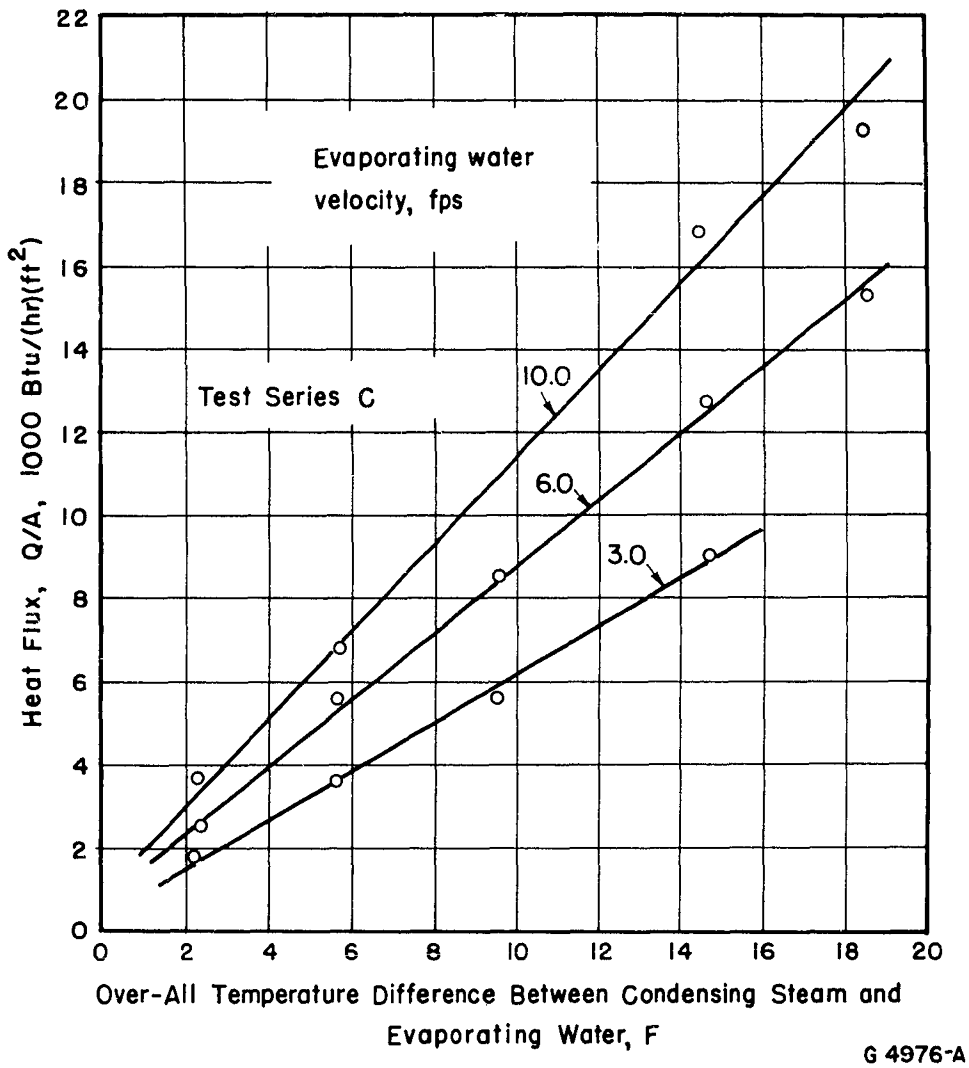


FIGURE 5. RELATION OF HEAT FLUX TO OVER-ALL TEMPERATURE DIFFERENCE AND EVAPORATING WATER VELOCITY FOR THREE 1/2-INCH-OD, 16 BWG BRASS TUBES, FILM CONDENSATION

difference, Δt, between condensing steam and evaporating water, is plotted. The Δt shown in the figures is the apparent temperature difference between the condensing steam and evaporating water. The term apparent temperature difference is descriptive in this instance because the temperature of the evaporating water is assumed to be equal to the saturation temperature of the steam, whereas actually this assumption is not exactly correct. Experiments by other investigators* have proven that in order for boiling to occur the temperature of the water must be slightly above the temperature of the steam. The amount of increase in temperature of the water above the temperature of the steam is dependent on the size of the steam bubble being generated. In addition, the water in the evaporator tubes is not at a uniform temperature throughout the tubes because the water is introduced at a temperature slightly below the steam point and rises in temperature as the water flows upward through the tubes. However, the apparent Δt, rather than the actual Δt, is the one of principal concern in designing thermocompression stills because the apparent Δt is, in effect, created by the steam compressor of the still.

The curves in Figures 3, 4, and 5 have, in all cases, been drawn as straight lines despite the fact that some of the points plotted, particularly those in Figure 4, indicate a slight upward curve.

Because all of the tests in a particular series were run with the same conditions of condensation, the curves reflect the improvement in heat flux obtainable with increases in evaporating water velocity and over-all Δt.

The velocities shown on the curves are the velocities of the evaporating water at the entrance to the evaporator tubes. The velocity at the tube exit is, of course, several times greater than at the entrance because of the large specific volume of the steam-water mixture at the exit. For example, at an inlet velocity of 10 fps with dropwise condensation and a Δt of 6 F the exit velocity from the tubes is approximately 39 fps, an increase of almost four times.

The results of the Series A and Series C tests were obtained with film-type condensation and the same conditions of forced-convection boiling. Therefore, the data of these tests reflect any changes in heat-transfer rates that might be attributed to a difference in the length to diameter ratios of the tubes. In the Series A tests, shown in Figure 3, the length to diameter ratio is 48:1 and in the Series C tests shown in Figure 5 the length to diameter ratio is 72:1. The curves in Figure 3 have a slightly steeper slope than those in Figure 5 but the order of magnitude of heat flux is the same in both cases at any given velocity and Δt. It is, therefore, indicated that the tube length to diameter ratio may have no significant influence on heat transfer when forced-convection boiling is used.

Condensing Coefficients

Figure 6 shows the relation of the heat flux for test Series A, B, and C to the condensing-film temperature difference. Examination of the curves in Figure 6 shows that in the range of film temperature difference of 2 to 3 F there is a 10 to 20 per cent increase in heat flux for film condensation over that predicted from the

*Jacob, Max, Heat Transfer, Volume 1, John Wiley and Sons, Inc., New York (1949), Chapter 29, Section 4, pp 620-624.

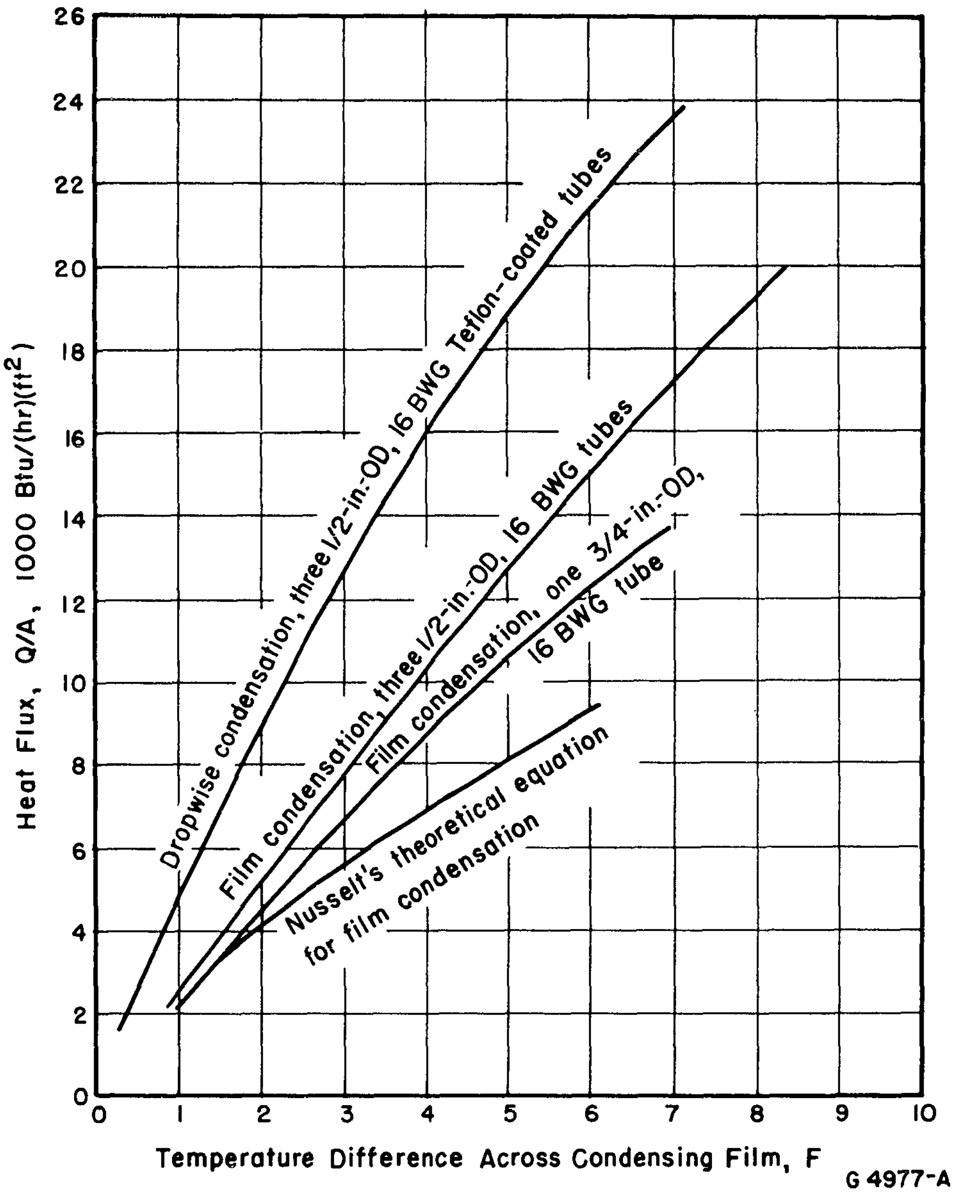


FIGURE 6. RELATION OF HEAT FLUX TO CONDENSING FILM TEMPERATURE DIFFERENCE

theoretical Nusselt curve. This increase over the theoretical is typical of that usually found with film condensation on a vertical tube and is explained by the fact that the film is turbulent as it flows down the tube whereas the theoretical curve assumes laminar flow.

The curve for dropwise condensation shows an improvement over film condensation of about two times at a 1 F film-temperature difference and 1.6 times at a 3 F film-temperature difference. The values shown include the resistance of the Teflon film used to promote dropwise condensation.

Evaporating Coefficients

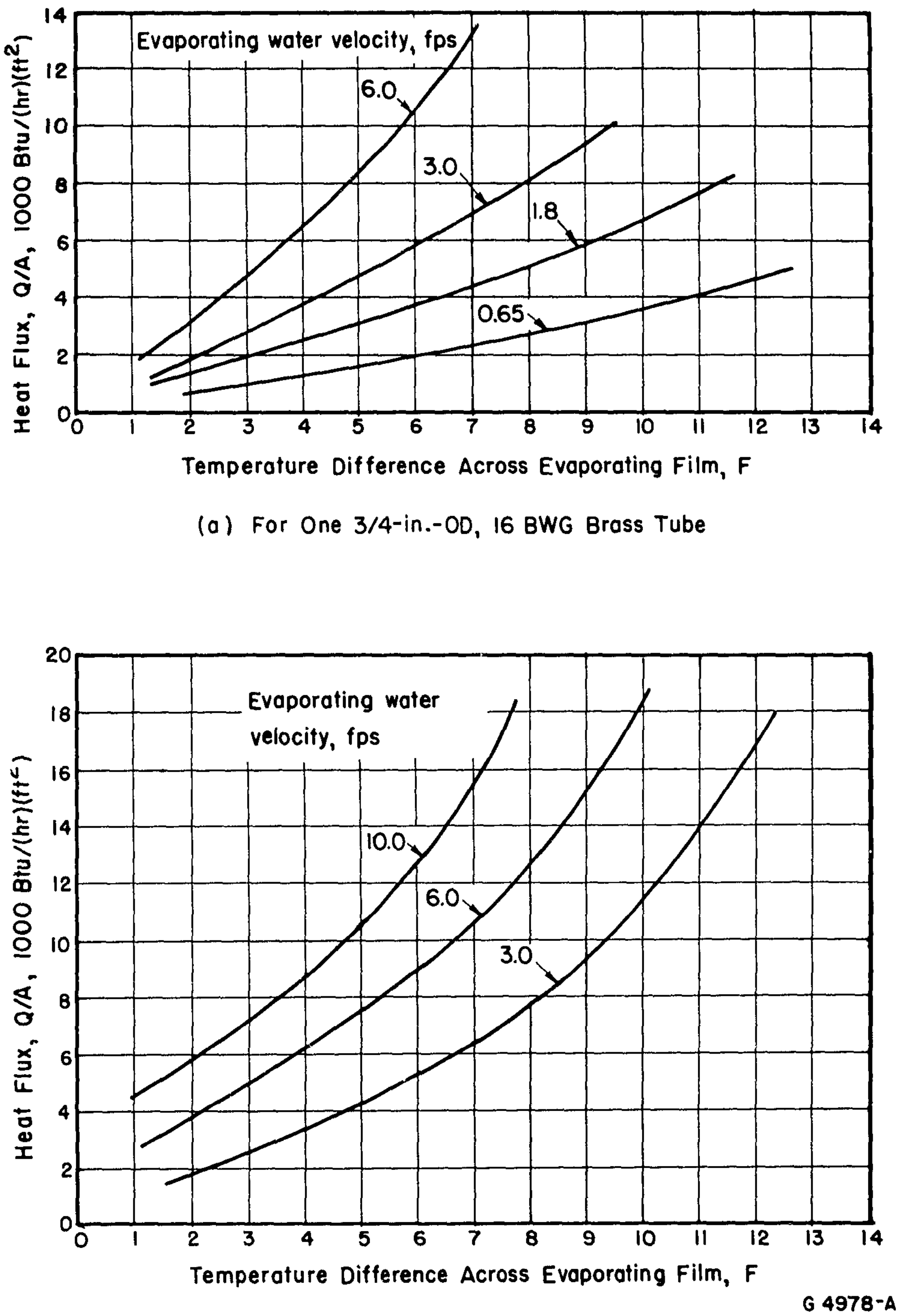
Figure 7 shows the relation of the heat flux to the evaporating film-temperature difference. The family of curves shown in the upper portion of the figure was computed from the Series A tests which used one 3/4-in. OD evaporator tube. The lower portion of the figure shows the evaporating heat-transfer rates from the B and C series of tests. The data on evaporating-film coefficient from the B and C series were combined because in these tests the tube size and evaporating water velocities were identical.

A comparison of Figures 6 and 7 shows that the evaporating film-temperature difference is the limiting factor on heat flux at water velocities of 3 fps and lower when film condensation is employed. At 6 fps, however, the condensing and evaporating film temperature differences are about equal. With dropwise condensation, the heat flux is controlled mainly by the evaporating film coefficient over the whole range of velocities tested.

Forced-Convection Flow Pressure Drop

Figure 8 shows the experimentally obtained pressure drop due to the flow of evaporating water through 1/2-in. OD, 16 BWG evaporator tubes. The curves show that the pressure drop increases with increases in Δt which is to be expected since the pressure drop is a function of both the tube wall friction and percentage of steam in the water leaving the tubes. At evaporating water velocities of 3 and 6 fps, the pressure drop due to evaporation of the flowing water is a larger part of the total drop than at 10 fps. Thus the former curves have a steeper slope. As a basis for comparison, the wall-friction pressure drop for water without boiling is 0.3-in. Hg, at 3 fps velocity, 1.0-in. Hg at 6 fps, and 2.4-in. Hg at 10 fps.

An attempt was made to evolve an equation which would correlate the measured pressure drop with the evaporating water velocity and the percentage of water evaporated. No equation was found which would reasonably satisfy most of the experimental data. The measured pressure drops deviated by ± 10 per cent from the curves shown in Figure 8. Additional tests over a wider range of velocity, temperature difference and heat flux, and with different sizes of evaporator tubes would be required before the pressure drop curves could be established with a high degree of certainty.



(a) For One 3/4-in.-OD, 16 BWG Brass Tube

(b) For Three 1/2-in.-OD, 16 BWG Brass Tubes

FIGURE 7. RELATION OF HEAT FLUX TO EVAPORATING FILM TEMPERATURE DIFFERENCE FOR ONE 3/4-INCH-OD, AND THREE 1/2-INCH-OD, 16 BWG BRASS TUBES

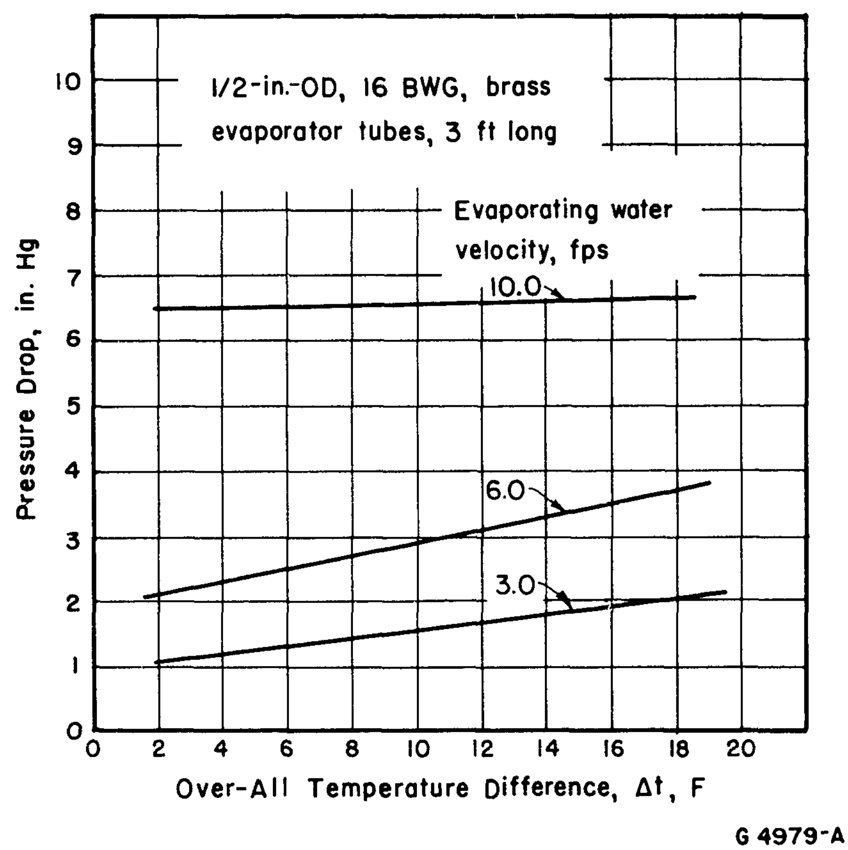


FIGURE 8. RELATION OF PRESSURE DROP INSIDE EVAPORATOR TUBES TO OVER-ALL TEMPERATURE DIFFERENCE WITH FORCED-CONVECTION EVAPORATION

REVIEW OF MATERIALS FOR SEA WATER EVAPORATORS AND HEAT EXCHANGERS

Sea water, with its high chloride-ion content, is corrosive to a great variety of metals. The rate of attack can be expected to increase rapidly as the temperature and rate of flow increase. Therefore, as the temperature is increased beyond about 120 F, the number of available metals with good corrosion properties becomes greatly reduced. The choice of metals for service in sea water at elevated temperatures, that is, up to 350 F, is, according to some experiments performed at the U. S. Naval Experiment Station, restricted to such materials as titanium, Hastelloy C (55 Ni, 17 Mo, 16 Cr, 6 Fe, 4 W), Inconel X (73 Ni, 15 Cr, 7 Fe, 2.4 Ti), and certain stainless alloys; however, for thermocompression stills, the selection is somewhat greater, since the maximum temperature is about 220 F.

Table 1 presents the results of corrosion tests of alloys noted for sea-water corrosion resistance. Undoubtedly, there are a few other test results available in the technical literature; however, it is safe to conclude that there is a paucity of information on the corrosive behavior of materials in hot, flowing sea water. Most of the test results are based on immersion in sea water at ordinary temperatures.

Experience has shown that, in general, copper-base alloys have the best resistance to sea-water corrosion. For velocities in the range of 2 to 6 ft per sec, aluminum brass (76 Cu, 22 Zn, 2 Al) is a good choice. For higher velocities, one of the cupro-nickels normally is found to be more suitable. An alloy containing 70 Cu, 30 Ni, 0.7 Fe is resistant to corrosion at high rates of flow. A less expensive alloy containing 89 Cu, 10 Ni, 1 Fe is considered almost as resistant at ordinary sea-water temperature. As shown in Table 1, the copper-nickel alloys give good service but tend to corrode locally at 350 F. However, at 220 F, there would be less tendency toward this type of attack.

Of the nickel-base alloys, Monel would be most likely to give good service at 220 F. A heavy corrosion scale is found on Monel after exposure at 350 F but, at 220 F, there would be less tendency for this to occur. Data are needed for both cupro-nickel and Monel at 220 F.

Of the stainless steels, those containing molybdenum are the most resistant to pitting attack in sea water. However, even the molybdenum stainless steels, such as 316 SS or Carpenter 20 are found to be rapidly attacked at local spots, e.g., under fouling or scale deposits. In Table 1, Type 316 SS showed good over-all performance, but there was local contact corrosion under the washers, where the test specimens were fastened to the fixture. Stainless steels with molybdenum have the advantage that they are much more resistant to general attack by high-temperature sea and fresh water and steam, than the copper or nickel-base alloys discussed above. Conditions must be controlled carefully to prevent fouling and local deposits if stainless steels are used.

Aluminum and certain aluminum alloys, such as those containing magnesium or silicon, have given reasonably good life in sea water at ordinary temperatures, however, it is not expected that aluminum would give good service in heated sea water, since it is inferior to copper and its alloys in sea water at normal temperatures.

TABLE 1. THE RESISTANCE OF SELECTED MATERIALS TO CORROSION BY SEA WATER

Material	Corrosion Rate, in./yr., From Weight Loss Test A (360 days at 70 F)	Test B (130 days at 70 F)	Test C (30 days at 350 F	Test D (54 days at 325 F)	Recommended Maximum Velocity of Sea Water, ft/sec	Remarks on Test C
Copper	0.0016	0.0011			3	
Red brass 85 Cu-15 Zn	0.0018	0.0013				
Admiralty (70 Cu-29 Zn-1 Sn)	0.0018	0.0012			3	
Aluminum brass (76 Cu-22 Zn-2 Al)	0.0008				7	
70-30 Cupro-nickel (0.7% Fe)	0.0003	0.0010	0.019	0.006	15	Localized corrosion
90-10 Cupro-nickel (1.7% Fe)			0.121	0.0015	15	Heavy corrosion scale
Monel (67 Ni-30 Cu-1.4 Fe)			0.031[a]			Heavy corrosion scale
304 SS (18 Cr-8 Ni)			0.103			Severe corrosion
316 SS (18 Cr-8 Ni-2.5 Mo)			0.00005			Slight contact corrosion
Titanium			G[a, b]			Stains at fixture contact

Test A Field test, one year in clean sea water at normal temperatures at 2-3 ft/sec, Kure Beach, N. C. Ref: "The Corrosion Resistance Characteristics of Copper and Nickel Alloys", H. O. Teeple, International Nickel Co., New York 5, N. Y.

Test B Field test, 130 days in Galveston Bay at a velocity of 1-2 ft/sec. Ref: Same as for Test A.

Test C Autoclave test with rotating sample holder providing a velocity of 10 ft/sec. Samples were exposed to fresh sea water, replaced every 15 days, at 350 F. Ref: U. S. Naval Experiment Station, "Testing of Various Materials in High Temperature Waters", EES Report 040028D, 30 November 1953.

Test D Autoclave test, 0.5 ft/sec, 54 days in 325 F sea water. Ref: Stewart and LaQue, Corrosion, Vol 8, No. 8, p 259-277 (August 1952).

(a) These samples were on test for 45 days.

(b) G = slight gain in weight due to stains at contact with fixture.

Bimetallic tubes are available from several manufacturers. Thus, it is possible to specify a copper-base alloy, such as 70 Cu-30 Ni on the inside of the tube and say aluminum on the other side. This should be a good combination for the evaporator tubes of a thermocompression still since the inside surfaces will be in contact with sea water, and the outside with nearly pure water or steam. Provided no heavy metals are dissolved in the steam condensate, aluminum can be expected to give good service. Bimetallic tubes, of course, cost appreciably more than single-metal construction.

The metal with the most outstanding ultimate promise for sea-water heat exchangers is titanium. Titanium, unlike other metals, normally does not pit, is not susceptible to stress corrosion, is free from local corrosion under fouling organisms, is free from impingement and cavitation attack at velocities which attack copper-base alloys, and is not susceptible to sulfide attack in contaminated sea water. Titanium and its alloys can be provided in the forms of sheet, tubing, or forgings. Some progress has been made in finding a method of producing coatings. Titanium and its alloys are less susceptible to scaling in sea water than other metals. Even though the thermal conductivity is low, the over-all efficiency is considered to be much greater in typical sea-water applications. The chief disadvantage of titanium is price, but some of this expense can be absorbed in the over-all cost of the equipment. In addition to titanium's corrosion resistance, it has the property that it is not wetted by water. Thus, its use in an evaporator would yield dropwise condensation and high heat-transfer rates.

In choosing materials for sea-water heat exchanger or evaporator service, one must also consider the forms available. Only materials available in wrought forms, such as tubing and sheet, have been discussed in this review. In all cases, the materials can be fabricated by usual methods including welding. Experience has shown that, while the relative costs of materials of construction may vary as much as 20 to 1, the finished installation at the site may only vary say 3 to 1. If titanium were used, for example, savings resulting from lower freight charges, reduced maintenance, longer service life, and greater over-all efficiency would at least partly compensate for the much higher initial cost of the metal.

Recommended Materials of Construction

At the present time, it is recommended that the heat-transfer surfaces of the evaporator and the heat exchangers be made of the 70 per cent copper, 30 per cent nickel alloy with 0.7 per cent iron. This alloy has given excellent service in heat exchangers aboard ship under a wide variety of service conditions.

Considerable research would be needed to demonstrate the possibility that useful life could be obtained by an all-aluminum design. Such a design probably could be evolved, but one would anticipate higher maintenance and replacement costs than for cupro-nickel alloy construction.

Titanium appears to warrant careful consideration as a material for the heat-transfer surface in evaporators of sea-water stills.

IMPROVED EVAPORATOR DESIGN FOR A 90-GPH THERMOCOMPRESSION SEA-WATER STILL

Two, basically similar, thermocompression evaporator designs are presented in this section of the report. The first design is based on the use of the smallest practical pressure difference between condensing steam and evaporating water. The second evaporator is designed to provide minimum heat-transfer surface without sacrificing greatly the economy or performance factor of the still. Both evaporators are designed to make use of dropwise condensation and natural convection evaporation. As is shown later, forced convection evaporation cannot be justified because the reduction in steam-compressor power associated with the higher heat-transfer rates possible with forced convection is more than offset by the additional pump power required for forced convection.

In order to establish a basis for the design of an improved evaporator, the evaporator presented in the summary report previously reviewed in the scope section of this report is used as the reference design.

Thermodynamic Design of Improved Evaporator

Any improvement in the performance of an evaporator must come from increases in heat-transfer rates at a minimum practical over-all temperature difference, Δt, between condensing steam and evaporating water. Heat-transfer rates can be increased by increasing the over-all Δt, but this method increases the power consumption of the still which results in lowered operating economy. As an alternative, the heat-transfer surface of the evaporator may be increased to provide the necessary heat transfer at a low temperature difference. A large evaporator permits a still design with good economy, but the evaporator becomes more expensive to construct and increases the size and weight of the still. It is thus apparent that a suitable design of evaporator must effect a compromise between minimum temperature difference and evaporator size. Moreover, the improvement in heat transfer must be accomplished without appreciably increasing power input for pumps or other fluid circulators.

The experimental phase of this project has been directed toward determining improved heat-transfer rates at the low over-all temperature difference necessary for efficient thermocompression still operation. The tests have shown that dropwise condensation and forced convection circulation both provide appreciably higher heat-transfer rates than have been possible before in thermocompression evaporator design.

Figure 9 is a plot of the heat-transfer curves that were used to estimate the performance of improved evaporator designs. The curves show the heat-flux at various over-all temperature differences for forced- and natural-convection boiling with dropwise condensation. Also shown for comparison is the case of natural convection with film condensation. The forced convection curves are based on experimental data except that they have been adjusted for the difference in heat flux between 1/2-in. OD, 16 BWG brass tubes used in the tests and 5/8-in. OD, 18 BWG 70-30 copper nickel tubes selected for the prototype evaporators. The natural-convection, dropwise-condensation curve was evolved by applying the dropwise condensing coefficients obtained during the present phase of the project to the natural convection heat-transfer curve presented in the previous Summary Report. The natural-convection, film-condensation curve was

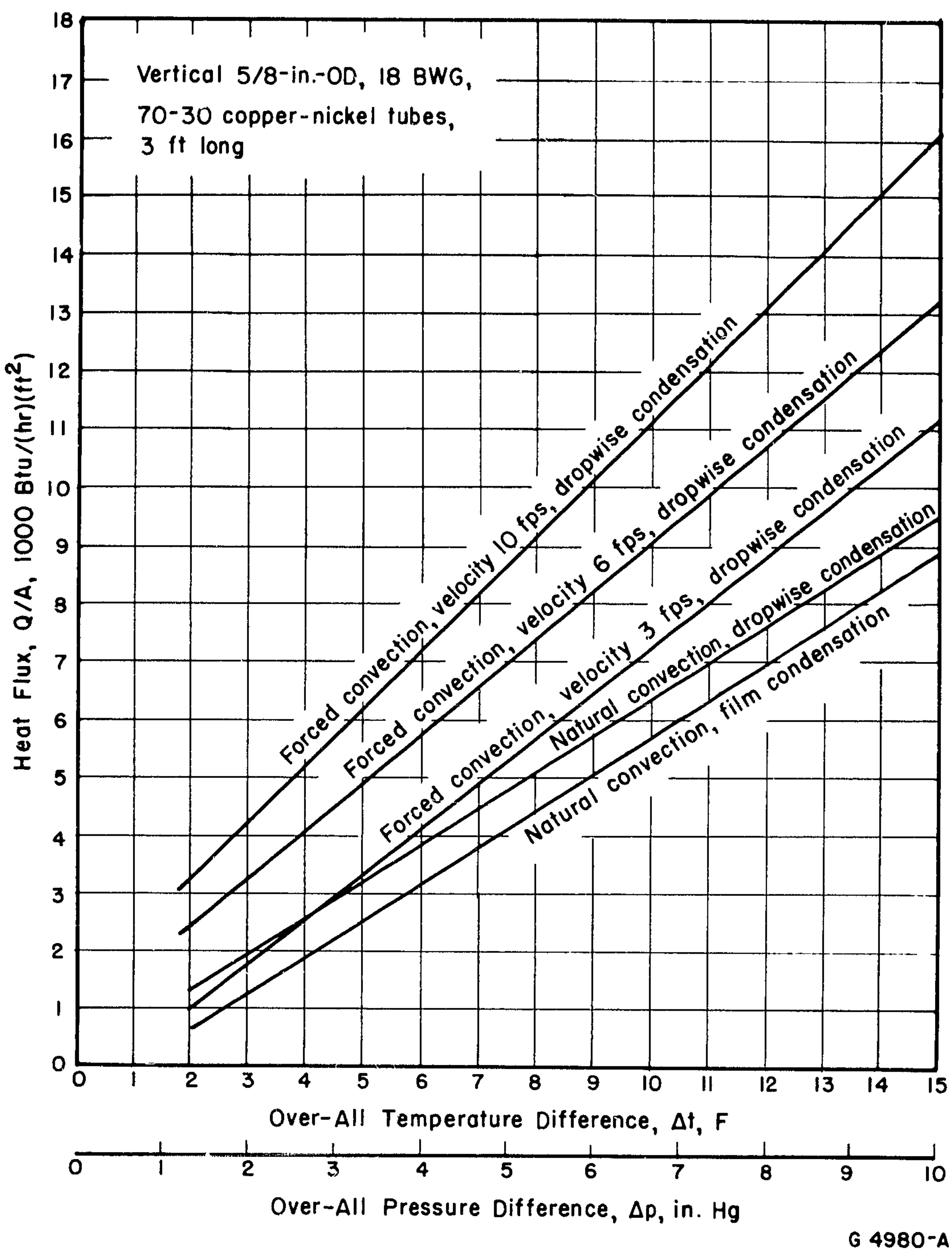


FIGURE 9. RELATION OF HEAT FLUX TO OVER-ALL TEMPERATURE DIFFERENCE FOR NATURAL- AND FORCED-CONVECTION EVAPORATION WITH DROPWISE CONDENSATION

obtained directly from the previous Summary Report. A comparison of the curves shows the magnitude of the improvement obtainable with dropwise condensation and forced convection evaporation. These curves form the basis for the coefficient of performance curves and the heat-transfer surface area curves shown in a subsequent figure.

Figure 10 shows the pump power required to circulate the evaporating water at 3, 6, and 10 fps, plotted against the over-all pressure difference between condensing steam and evaporating water. The pump power is based on the experimentally determined pressure drop data shown in Figure 8 and on the mass rate of water flow required to produce the forced convection evaporating water velocity. The mass rate of flow is dependent not only upon the velocity but also upon the number of tubes required at any particular pressure difference. The curves turn upward at the lower values of Δp. The reason for this is that in this range of Δp the heat flux is low requiring a large number of tubes and consequently a high mass rate of water flow to produce the required velocity.

The pump for a forced-convection evaporator could be of the propeller type and could be located in the downcommer of the evaporator. The pump shaft could be extended through the bottom head of the evaporator by means of a packing gland and driven by a V-belt. The efficiency of a pump of this type has been conservatively estimated at 40 per cent and this value was used in computing the curves.

Figure 11 shows two families of curves which are based on the experimental data shown in Figures 8, 9, and 10 and on the performance characteristics of the still components that were reported on in the previous Summary Report.

These performance characteristics are:

(1) Steam compressor adiabatic efficiency, 60 per cent

(2) Diesel engine specific fuel consumption, 0.5 lb/bhp-hr

(3) Pressure on evaporating side of evaporator, 32-in. Hg

(4) Ratio of blowdown to feed, 2:5

(5) Boiling point elevation of evaporating water, 1.5 F

(6) Total power required for distillate, blowdown, and feed pumps, 0.5 bhp

(7) V-belt drive efficiency, 97 per cent.

Since the performance characteristics of the still components in Figure 11 are the same as for the hypothetical still, all the improvement in the performance factor of the still results from improved heat transfer in the evaporator. It should also be emphasized that the curves of Figure 11 are based on one specific set of design conditions. Changing the design conditions would affect the magnitude of the values of the curves but this would not alter the trends indicated by the curves.

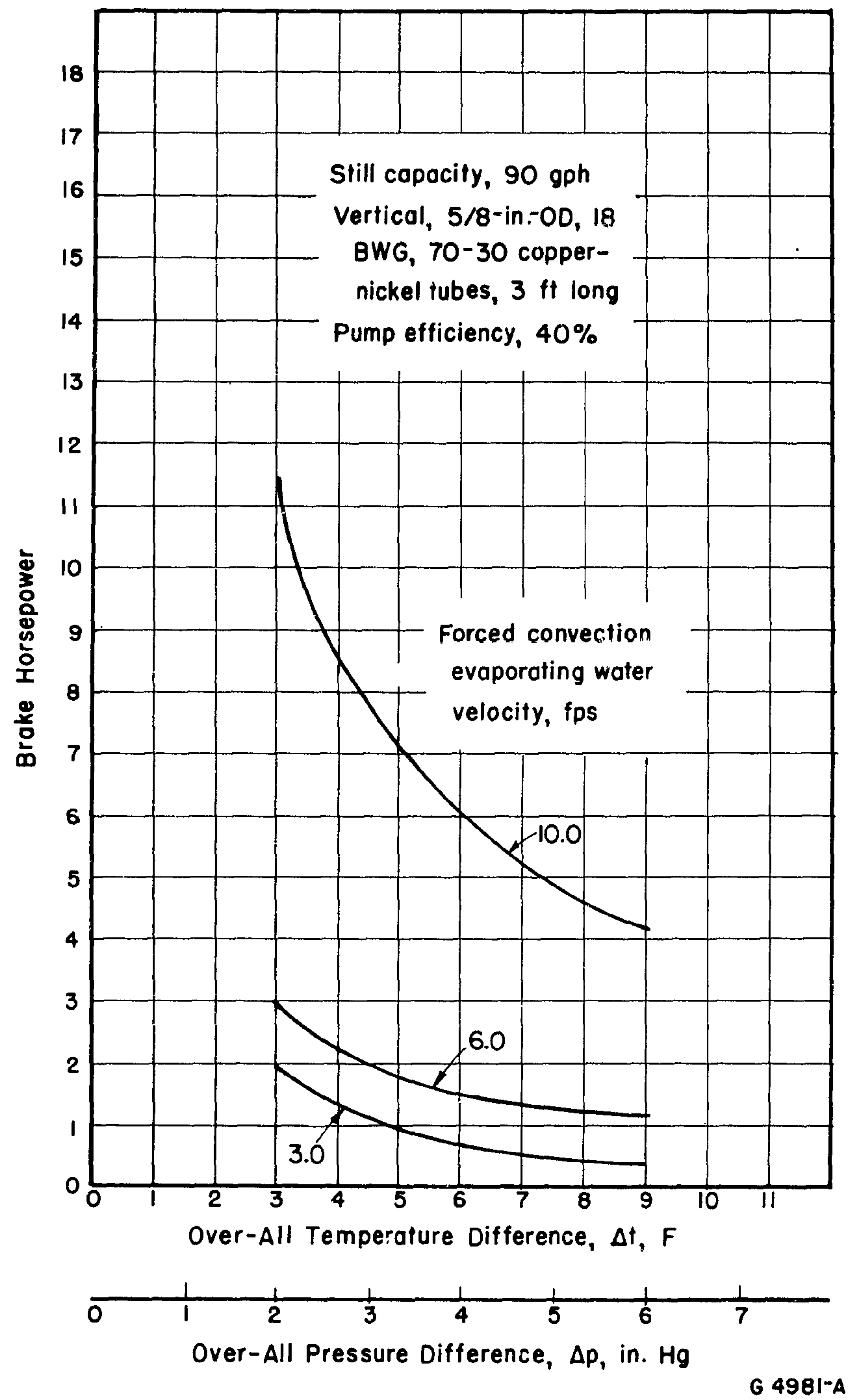


FIGURE 10. RELATION OF PUMP POWER REQUIRED FOR CIRCULATING EVAPORATING WATER TO TEMPERATURE AND PRESSURE DIFFERENCE

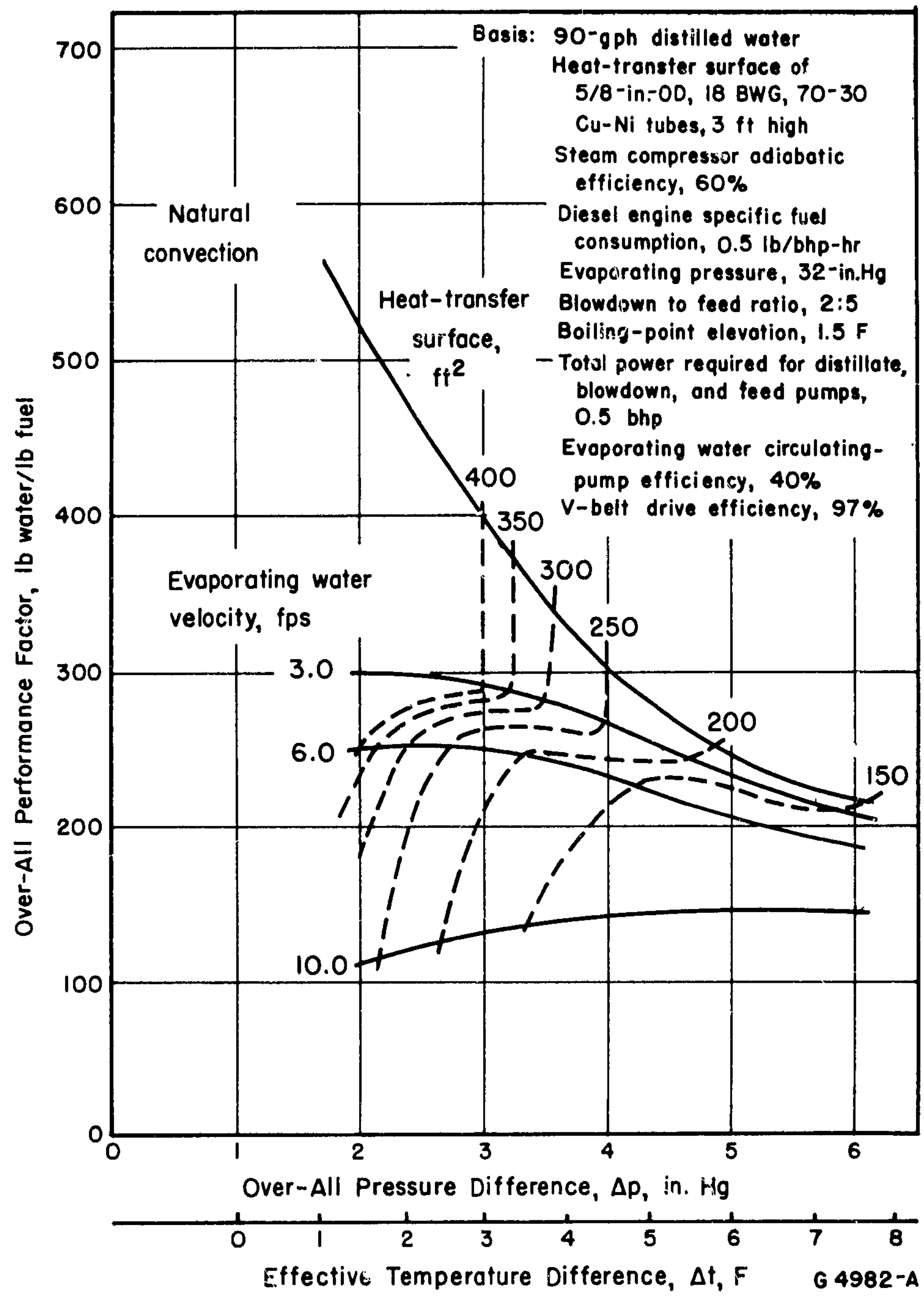


FIGURE 11. CURVES SHOWING TRENDS OF OVER-ALL PERFORMANCE FACTOR AND HEAT-TRANSFER SURFACE REQUIREMENTS FOR FORCED- AND NATURAL-CONVECTION EVAPORATORS UTILIZING DROP-WISE CONDENSATION

One family of curves in Figure 11 gives the performance factor for a 90-gph still at various values of over-all pressure difference and at various evaporating water flow velocities. The second family of curves shows the evaporator surface area required to evaporate 90 gph of salt water. The points of intersection of the two sets of curves establish the evaporator size and performance factor for any given pressure difference. For example, a still equipped with an evaporator having 250 sq ft of heat transfer surface and operated at a forced convection velocity of 3 fps and a pressure difference of 4.00-in Hg would have an over-all performance factor of 265 lb of distilled water per lb fuel.

Examination of the curves in Figure 11 shows that the best performance factor is obtained with natural convection. With forced convection, the performance factor decreases as the forced convection velocity increases in spite of the fact that the heat-transfer rates increase as velocities increase. This decrease in performance is explained by the fact that the pump power required to circulate the water in forced convection is not compensated for by the reduction in compressor power made possible by the increase in heat-transfer rate.

Based on the curves shown in Figure 11, it appears that an evaporator utilizing dropwise condensation and natural convection evaporation provides the best performance factor for a still having a reasonable heat-transfer surface area. The minimum practical pressure difference for stable operation of a small evaporator is believed to be of the order of 3-in. Hg. An evaporator operating at 3.25 in. Hg pressure difference would require 350 sq ft of heat-transfer surface to produce 90 gph of distilled water. A still using this improved evaporator would have a performance factor of 370 lb distillate per lb of fuel. The evaporator used as a basis for comparison operated at a 4-in. Hg pressure difference, had 320 sq ft heat-transfer surface and had a performance factor of 300 lb distillate per lb fuel. By making use of the increased heat transfer rates possible with dropwise condensation the improved evaporator design gives approximately a 23 per cent increase in performance factor with only a 10 per cent increase in surface area.

As an alternate, an evaporator that would permit a performance factor of 300 lb distillate per lb fuel, the same performance as is used as a standard for comparison, could be designed. This evaporator would need only 250 sq ft of heat-transfer surface compared to 320 sq ft for the standard unit, or a reduction of about 20 per cent.

Table 2 gives a summary of the performance characteristics of the two improved evaporators discussed in this report and of the performance of the evaporator presented in the previous summary report.

TABLE 2. SUMMARY AND COMPARISON OF EVAPORATION PERFORMANCE

	Previous Design, Natural Convection, Film Condensation	Improved Design for Best Operating Economy, Natural Convection, Dropwise Condensation	Improved Design for Minimum Surface Area, Natural Convection, Dropwise Condensation
Performance Factor, lb distillate per lb fuel	300	370	300
Surface Area, ft^2	320	350	250
Number of 5/8-In. OD, BW 6 tubes	650	714	510
Approximate Shell Diameter, in.	25	27	24
Operating Pressure Difference, in. Hg	4	3.25	4
Heat-Transfer Coefficient, Btu/(hr)(ft^2)(F)	525	645	635
Rated Still Output, gph	90	90	90

Mechanical Design of Improved Evaporator

The principal mechanical design problem associated with an evaporator is that of fabricating a durable unit at minimum cost. Because all of the components in contact with salt water must be corrosion resistant and all of the parts in contact with the distillate must not contaminate the distillate, a copper-base alloy is recommended for all parts of the evaporator. The tubes, tube sheets, and bottom head, because they are in contact with flowing, concentrated sea-water, should be fabricated from a 70-30 copper-nickel alloy as is pointed out in the materials section of this report. The shell and top head of the evaporator should be constructed from a copper-base alloy. It is recommended that the evaporator be made in three sections, the bottom head, tube bundle and shell, and top head joined with bolted flanges, and that all external pipe connections be either standard threaded pipe joints or bolted flanges.

Figure 12 shows the general configuration and over-all dimensions of the improved evaporator. The 340-sq ft and 250-sq ft units are the same except for the number of tubes and shell diameter.

The steam separator shown in Figure 12 is of the type made by the Otto H. York Company, Inc., of East Orange, New Jersey. The separator consists essentially of a strip of woven wire mesh formed into a pad and supported on a lightweight grid. In operation, the steam-water mixture impinges on the mesh pad. The steam passes through while the water collects on the mesh and drips off. The manufacturer has recommended a separator 8 in. thick in their style No. 421 high-efficiency mesh. The unit would be fabricated in two 4-in. thick layers with each layer further subdivided into two sections. Each section in each layer would be supported by a lightweight grid. The

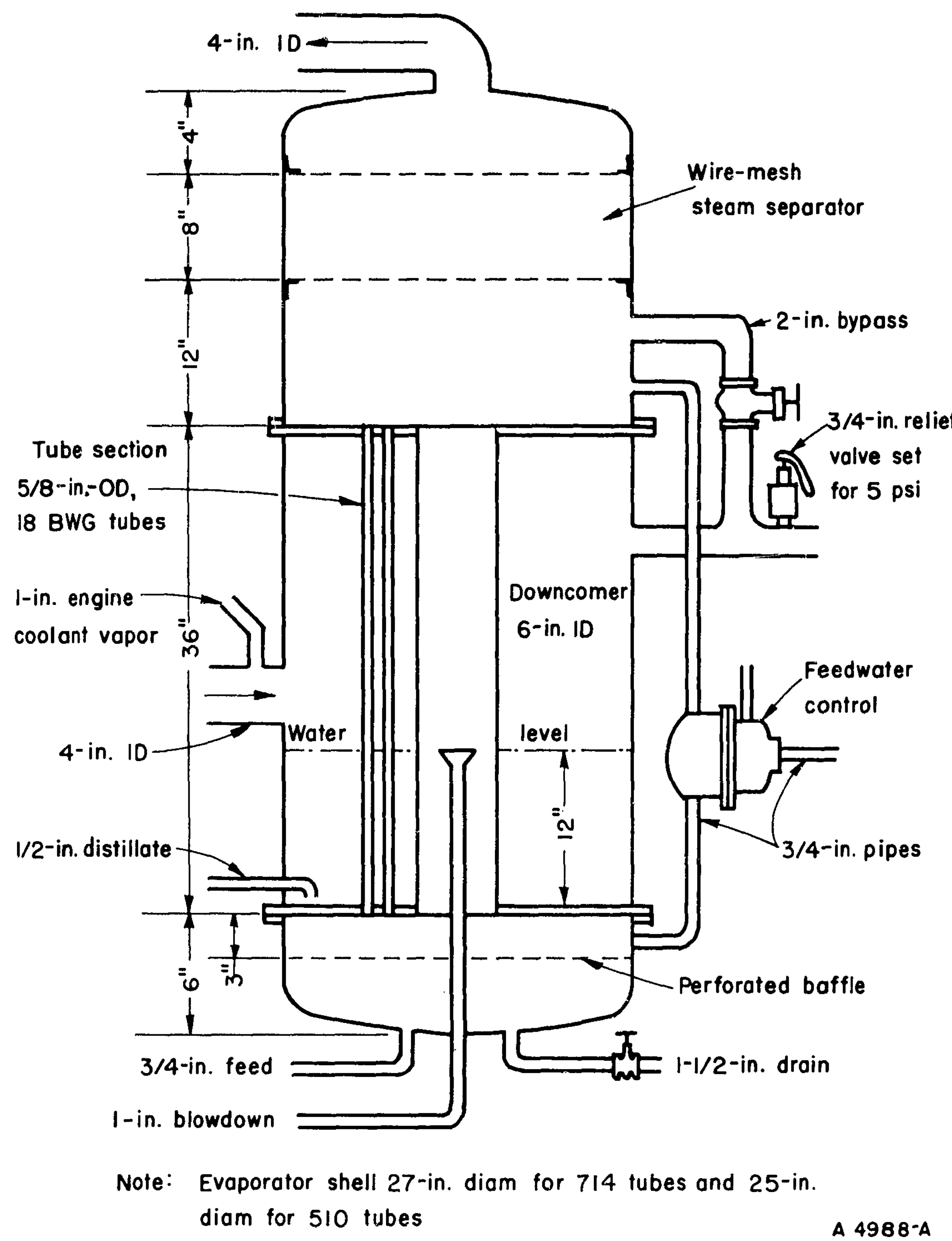


FIGURE 12. IMPROVED EVAPORATOR FOR 90-GPH STILL

SUGGESTED FUTURE WORK

During the course of this project several areas for additional work that could lead to further improvement in thermocompression still performance have been recognized. Inasmuch as the improved evaporator performance developed under this project has been determined with laboratory equipment, it is recommended that a full-size evaporator be fitted with Teflon coated tubes and operated on a thermocompression cycle with sea-water to verify the performance predicted by the laboratory experiments. In addition to verifying the improved design, tests on a full-scale evaporator would permit service life studies of the Teflon coating on the tubes.

Because the Teflon film used for dropwise promotion offers some thermal resistance, heat transfer studies using other means of promoting dropwise condensation should be considered. Other coatings, besides having lower thermal resistance than Teflon, may also have a larger contact angle between the water drop and the tube surface thereby permitting even higher dropwise condensing coefficients. Certain silicone compounds are extremely nonwettable and could be applied to tube surfaces in molecular layers to promote dropwise condensation. Titanium is not wetted by water and could be expected to be a dropwise promoter. Titanium also shows almost no corrosion in sea-water and resists scaling better than some other materials. It is therefore believed that additional research in this area would be beneficial.

Additional studies similar to some of those just completed could be expected to produce gains, although perhaps small, in thermocompression still performance. Further study of the mechanics and thermodynamics of natural convection may lead to improvements in downcomer design and tube arrangement for natural convection evaporators. Research to determine the optimum tube-length to diameter ratio could also lead to improvement in evaporator performance. Drip dams applied to the outside surface of evaporator tubes are worthy of further study as their use should lead to higher condensing coefficients. Drip dams, it is expected, would show improvement with either film or dropwise condensation. Lastly, studies relating surface roughness to both natural- and forced-convection evaporation might result in better boiling film coefficients.

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Data upon which this report is based may be found in Battelle Laboratory Record Book No. 11959.

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